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HARMONIC VIBRATION TESTING OF A NON-LINEAR AIRCRAFT STRUCTURE

C. Beatrix

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ABSTRACT. Realistic vibration testing methods for aircraft structures in the presence of dry friction are discussed.

I. INTRODUCTION

Structural non-linearities of aircraft, encountered chiefly in the steering /58* gear of light planes, and their effects on prediction of the critical velocity of aeroelastic instability have already been treated in several papers in La Recherche Aerospatiale [1, 2, 3].

These non-linearities are produced essentially by "dry" friction in the steering gear of the control surfaces. The different "thresholds" which result have the effect of a conspicuous non-linear stiffness in the drive of the control surfaces. Thus, during harmonic vibration testing it is difficult to obtain a homogeneous linear configuration, and even if that difficulty is resolved, it is not certain that such a configuration will represent the aircraft with the in-flight load factor — for example, deformation of a control surface by a static load can "line up" the bearings, and so suppress some of the friction. Consequently, the harmonic vibration test must characterize the structure not only as it is at the time of manufacture, but also in the "frictionless" state which it will reach either under load, or after a certain number of hours of flight time.

* Numbers in the margin indicate pagination in the original foreign text.

To solve this double problem the vibration test must be run on a structure which has been modified so that the dry friction is negligible. Sometimes this can be accomplished by minor technical modifications, but in most cases these modifications, in addition to being easily reversible, must be effective, for example:

- loosening the control surfaces from their steering gear;
- locking the kinematics of the control surfaces;
- modifying the balance of the control surfaces;
- superposing model characteristics of the structure without control surfaces and steering gear ("branch-mode" method).

The first solution is the most simple technically, but it is sometimes not very effective, especially for friction localized in the bearings or in the hyperstatic linkages. The last two solutions, aside from being complicated, make interpretation difficult.

This is why the second solution is often used.

It can be done relatively simply, and so there is no particular difficulty in returning the structure to its initial state, or in determining the ideal "frictionless" structure. Nevertheless, there have sometimes been errors in performance and interpretation. For this reason, it is useful to define the circumstances of its application by an example representing a steering gear for an elevator control system.

II. REPRESENTATION OF A STEERING GEAR WITH TWO DEGREES OF FREEDOM. LOCKING OF ONE DEGREE OF FREEDOM.

Let a representation of an elevator control system (Figure 1) consist of:

- a control surface with inertia I_1, turning about an axis 0_1 fixed with respect to the structure; position variable X_1; on this axis is friction which appears as a stiffness torque C_0;

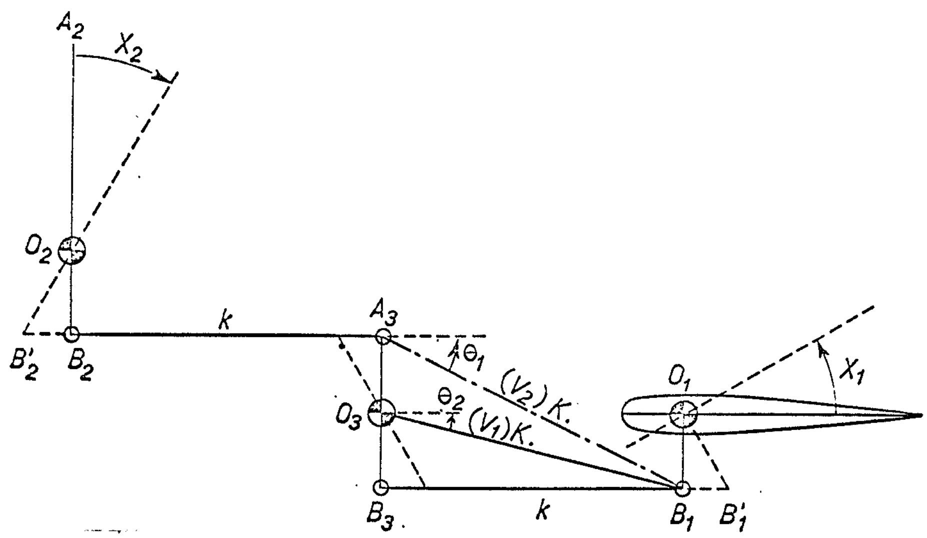

Figure 1. Diagram of a control system for an elevator control system.

This error approaches zero when η (i.e., the stiffness K) approaches infinity; λ_1, and λ_2 then have the limits $\frac{1}{2\xi}$ and infinity, respectively.

Case V_2

In this case, the smallest error with respect to the frictionless system occurs in the harmonic ε_2; this error approaches a finite nonzero value when K approaches infinity:

$$\varepsilon_2 \longrightarrow \frac{\xi\eta_0(1+\eta_0)}{\xi(\eta^2+4A^2)+\eta_0(1+2\xi)}.$$

III. III. CONCLUSION

Although the preceding example is highly schematic, it shows the advantages of Case V_1 over Case V_2.

It is seen that in the first case, the relative error in the fundamental frequency with respect to the frictionless system approaches zero as $\frac{1}{\eta^2}$

for large values of η, whereas the smallest error in the second case (ε_2) approaches a nonzero value.

Consequently, locking the steering gear to a fixed point of the structure is more effective and more accurate, and accomplishes the purpose of making the parasitic stiffness due to friction negligible with respect to the modification of the stiffness introduced by the supplementary connecting rod.

Whatever the solution adopted, it is necessary during the harmonic vibration test to know both the stiffness K of the locking rod (which can be obtained by a very simple preliminary test) and the displacements δ_i of the ends of the rod in different modes. This will permit correcting the generalized stiffness matrix $[\gamma]$ by a matrix $-K[\Delta]$, of which the i, j term is $-K\delta_i\delta_j$, and thus the proper modes of the frictionless steering gear can be determined.

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